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AN EXPERIMENTAL INVESTIGATION OF TURBULENCE—EXCITED PANEL VIBRATION AND NOISE (BOUNDARY—LAYER NOISE)

by

M. Y. EL BAROUDI, G. R. LUDWIG and H. S. RIBNER

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SUMMARY

A detailed study has been made of the flexural motion and noise generated by 11 × 11 inch steel panels flush-mounted in the wall of a turbulent flow channel. The mean square exciting pressure fluctuation at the wall, its spectral density, and two-point correlations of the pressure were measured with the use of pinhole microphones.

The flexural response of sample panels was studied by correlation techniques. The calculated relief plot of correlation shows qualitative agreement with the experimental results.

SOMMAIRE

On a procédé à une étude détaillée du mouvement de flexion et du bruit engendrés par des panneaux métalliques de 28 cm x 28 cm, montés affleurants dans la paroi d'un canal d'écoulement turbulent. A l'aide de microphones à trou d'épingle, on a mesuré la moyenne des carrés de fluctuation de pression excitante à la paroi, sa densité spectrale et les corrélations à deux points de la pression.

La réponse de flexion des panneaux échantillons a été étudiée par les techniques de corrélation. Le tracé graphique de relief calculé de corrélation dénote une concordance qualitative avec les résultats expérimentaux.

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NOTATION

d	diameter of pressure transducer sensing element
db	decibel (see, e.g., Fig. 17)
e_1, e_2	two arbitrary functions of time
f	frequency in cycles/sec
h, h_0	panel thickness
L	panel length
M	Mach number
$p(f)$	mean square wall pressure per cycle
q	duct center line dynamic pressure
r.m.s	root mean square
R	correlation of measurements at two points with time delay
U, U_0	duct center line flow velocity
U_c	convection velocity of driving pressure
$\bar{U}_c$	average value of measured driving pressure convection velocities
U_p	speed of panel flexural wave
z	vertical height above test panel
γ	$\cos^{-1} (\lambda/\lambda_x)$
δ	half depth of air duct (equivalent boundary layer thickness)
δ^*	displacement thickness
Δx	separation distance between two observation points in flow direction
ξ	critical damping ratio
λ	wave length of the panel flexural wave
λ_x	wave length of a Fourier component of the driving pressure
τ	time delay
ω_{mn}	panel modal frequency (m,n are the number of loop lines defining a particular mode)

AN EXPERIMENTAL INVESTIGATION OF TURBULENCE-EXCITED PANEL VIBRATION AND NOISE (BOUNDARY-LAYER NOISE)*

M.Y. el Baroudi**, G.R. Ludwig† and H.S. Ribner††

1. INTRODUCTION

There are two mechanisms by which a turbulent boundary layer can create noise as it passes over a solid surface. If the surface is rigid, the turbulent pressure fluctuations in the flow radiate sound directly into the air adjacent to it. If the surface is flexible, the turbulent boundary layer passing over it excites vibrations in the surface. The vibrating surface then acts as a radiator of sound similar to the diaphragm of a loudspeaker. Both mechanisms are present for a flexible wall, but the second may be more efficient for thin walls at subsonic flow speeds[1].

The results reported herein have been compiled from two separate, although complementary, investigations. The first investigation was designed to study experimentally the effect of various flow and plate parameters on the total sound power radiated by a thin flexible panel when excited by a turbulent flow. The parameters chosen as variables were flow velocity, boundary-layer thickness and panel thickness. In addition, the total damping, although not systematically varied, was measured for sample panels of each thickness mounted in their test environment. Fully developed turbulent channel flow was chosen as a source of turbulence. Measurements of the mean square pressure fluctuation at the wall, its spectral density and two point space correlations of pressure were made in order to specify in a limited fashion the forcing function acting on the panels.

The second investigation was initiated to study experimentally and theoretically the motion of the surface of a panel excited by turbulent flow. The physical environment was identical to that used in the first investigation. The statistical quantities measured were the two point space-time correlation of the turbulent wall pressure field and the two point space-time correlation of the resultant panel surface motion as well as its magnitude.

2. THEORY

A body of theoretical investigations of the surface motion and attendant noise radiation of flexible panels excited by turbulence is to be found in the literature[2-8]. Two levels of approximation have been used. Ribner[2], whose work is representative of the simpler approach, idealized the flexible surface as an infinite panel. He treated the response in terms of running waves which were excited by a 'frozen'

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convected pattern of turbulence moving over one side of the panel surface. Dyer[3], whose work typifies the second approach, considered the motion of a finite plate excited by a decaying convected pressure pattern in terms of its normal modes.

The theoretical treatment of the panel surface motion conducted in conjunction with the experimental investigation was based on the method of Dyer. The problem considered was the response of a finite, simply supported plate excited by a wall pressure field whose idealized space-time pressure covariance was expressed in terms of a delta function approximation. The use of such an approximation effectively subjected the panel to a point source loading rather than a distributed loading. Approximate numerical values for the quantities needed in this representation were found experimentally from the turbulent channel flow. The particular form of the response sought was the displacement covariance. This was expressed in terms of the plate impulse response and the pressure field covariance. The integrals of the displacement covariance and its Fourier transform, the power spectral density function, were evaluated completely for this investigation.

In evaluating the response, the back reaction of the panel motion on the turbulent flow was assumed to be negligible. This seemed plausible because the root-mean-square displacement was small, the ratio of the boundary-layer displacement thickness to the r.m.s. panel displacement was approximately 2,000. Moreover, unpublished estimates of the back reaction by Ribner in connection with his work on panel radiated sound indicated such a conclusion.

When a driving pressure wave matched a possible plate flexural wave in speed and in wavelength, a condition similar to resonance, called coincidence, was met. This is sketched in Figure 1. In the Figure, U_c and λ_x are the convection velocity and wavelength of the driving pressure, and U_p and λ are the speed and wavelength of the flexural wave. The relationship for angular frequency, ω_{mn}, gives a maximum frequency for coincidence. The upper limit to the coincidence frequency range was taken to indicate that only a finite number of modes were necessary to describe the spectrum of the panel displacement. This result was also derived by Ribner for an infinite panel.

The power spectral density function and longitudinal two-point space-time correlation function of the panel displacement were evaluated on an IBM 7090 digital computer. Some of the results are presented in Section 4.3 for comparison with the experimentally determined properties of the panel surface motion.

3. EXPERIMENTAL APPARATUS AND PROCEDURE

The facility for generating turbulent channel flow was basically an open circuit wind tunnel with interchangeable rectangular duct sections twelve inches wide and one or eight inches deep. The test section of the duct is sketched in Figure 2. The interiors of the ducts were lined with acoustic tile in order to attenuate any noise originating at the centrifugal fan used to drive the tunnel. Panels to be tested were fitted in a hole in the top of the duct, flush with the inner surface.

The panel material was commercial shim steel. Four thicknesses were used in the noise tests: 0.0015, 0.002, 0.004 and 0.008 inches. Only the 0.002 and 0.008 inch

panels were used in the panel surface motion experiments. The shim steel was mounted in a steel frame with all four edges clamped, leaving an exposed area of 11 x 11 inches. Care was taken so as not to introduce any appreciable amount of tension.

Both ducts were sufficiently long upstream of the test section to create fully developed turbulent flow at the panel. A diffuser situated downstream of the test section was designed to give a static pressure in the duct which was slightly less than atmospheric at the panel station. This was done to prevent in and out movement of the panel, best described as an oil-can motion, with slight variations in atmospheric pressure. Conventionally designed total and static pressure probes were used to measure the velocity profiles at various stations in each duct.

Acoustic measurements were made in a double-walled reverberant room surrounding the panel. The panel was situated near one end of the irregularly shaped room and a condenser microphone at the opposite end detected the sound pressure level. The microphone signal, after amplification and filtering into 1/3 octave bandwidths, was read on a Flow Corporation Random Signal Meter. The latter instrument reads true root-mean-square of any random signal and has a 16-second time constant. The reverberant room was calibrated by using a sound source of known characteristics to provide the proportionality constant connecting acoustic power input in 1/3 octave bandwidths to the average sound pressure level measured by the microphone. The known source was essentially a stalled centrifugal fan (I.L.G. Standard Sound Source, Model DSN 10910)[9].

Background noise levels were found by replacing the panel with a wooden plug, 1½ inches thick. The level was found to be acceptably low for all but the thickest panels. For the latter, a background correction was required. Setting the panel in its frame on top of the wooden plug failed to increase the measured background noise; this showed that duct vibration was not adding to the panel generated sound. Because of a dominance of background noise below 100 cycles/sec, a high pass filter was used to limit all broad band measurements to frequencies above this value.

The technique used to measure damping was the resonance or bandwidth method. The panel, mounted in the 8-inch duct, was excited by an electromagnet, and its response was sensed with a capacitance displacement probe of the Shattuck type. The critical damping ratio of the panels at any resonant frequency was computed from the frequency bandwith measured 3 db below the peak of the resonant displacement versus frequency curve. The input frequency to the magnet was resolved to within approximately ± 0.01 cycles/sec between 20 and 200 cycles/sec and ± 0.1 cycles/sec between 200 and 2000 cycles/sec by means of a Lissajou figure technique involving an oscilloscope and a wave analyser[10]. The presence of more than one mode of vibration in the panel was readily apparent on a monitor oscilloscope. Any such multiple response was rejected along with modes for which frequency separation was less than 5 times the measured bandwidth.

The spectra of the turbulent wall pressure fluctuations in the ducts were measured with a pinhole microphone supplied by Altec Lansing. There was a single hole set near the edge of the microphone face. The diameter of the hole was 0.04 inch. An equalization filter network was added to the electronic equipment used in the acoustic measurements in order to compensate for mechanical resonances of the pinhole microphone

at 5,000 and 11,000 cycles/sec. The microphone was mounted with its face flush in a frame designed to be interchangeable with the panel frame.

Two point space and space-time correlations of the pressure fluctuations were obtained by placing two pinhole microphone systems at various separations in the frame and feeding the resulting pressure signals into an electronic correlator. High pass filters with very sharp cutoffs at 100 cycles/sec were used to exclude spurious signals of acoustic origin.

Measurements of the panel transverse displacement were made with Shattuck type capacitance probes whose sensing elements were .125 inch in diameter. Investigation of the frequency response of probes of this general type by Shattuck[11] and by Koidan[12] indicates a flat response up to 10,000 cycles/sec. The correlation measurements involved two such probes which were mounted in a two-degree of freedom traversing mechanism at a height of less than .1 inch above the panel (Fig. 2). The electronic equipment was identical to that used with the pinhole microphones with the exception of the microphone equalization filters.

The electronic correlator is shown schematically in Figure 3. The heart of this system was an Ampex dual channel tape recorder with a special staggered head assembly. The staggered head was capable of introducing time delays, τ, of up to 30 milliseconds between two signals, e_1 and e_2, which were originally recorded simultaneously. The two signals, one of which was delayed in time with respect to the other, were then fed through a switching box to a Philbrick analog multiplier. The switching box was used to select the instantaneous product of both signals or the square of either signal as outputs from the multiplier. When only limited portions of the frequency spectra were of interest, the signals were passed through two identical Muirhead Pametrada wave analysers before feeding them into switching box.

The multiplier output was averaged by an active resistance capacitance integrator whose integrating time was 10 seconds. The normalized correlation function of the two signals was then computed from the three output readings available at the integrator. These were the time average of the product e_1e_2 and the time average of the squared signals e_1^2 or e_2^2. The system whose frequency response extended from 18 to 10,000 cycles/sec accurately autocorrelated sine waves over this frequency range.

4. EXPERIMENTAL RESULTS AND DISCUSSION

4.1 Wall Pressure Field

Velocity profiles in both 8- and 1-inch ducts were measured upstream and downstream of the panel station in order to ascertain the fully developed nature of the turbulent channel flow. The velocity profiles so obtained were identical for all stations in both ducts when nondimensionalized by the channel half-depth and the centerline flow velocity. The effective displacement thicknesses of the turbulent flow were calculated from the velocity profiles. The numerical values were 0.0362 ft and 0.00453 ft for the 8- and 1-inch ducts respectively.

The spectral densities of the wall pressure fluctuations were calculated from the 1/3 octave analyses of these fluctuations. The result for the 8-inch duct is shown

in Figure 4. The data scaled roughly when plotted in the nondimensional form $p(f)U/q^2\delta^*$ versus $f\delta^*/U$. In this notation $p(f)$ is the mean-square pressure per cycle, q and U the dynamic pressure and velocity at the duct centerline, and δ^* the displacement thickness. The average spectral density curve obtained in the 1-inch duct has been added as a solid line for comparison. Note that the solid line indicates a levelling out of the spectral density curve below a nondimensional frequency of approximately .04. This flat portion of the spectrum was found to extend down to the lowest value $f\delta^*/U$ measured, which was .0048.

The sharp drop in the spectral density curves, above $f\delta^*/U = 2$ for the 8-inch duct and 0.3 for the 1-inch duct, can be explained as follows. The microphone response falls off markedly for turbulent eddy sizes comparable with or smaller than the pinhole diameter. These smaller eddies correspond to the higher frequencies in the convected turbulence; hence, the sharp drop above some critical frequency. Corcos et alii[13] have developed a relation between the measured and true spectral density as a function of fd/U_c, where d is the transducer diameter and U_c is the convection speed of the pressure producing eddies. This correction was calculated for the two ducts and the cutoff frequency was taken as the frequency at which the correction reached 1 db. These calculated cutoff frequencies are shown in Figure 4 for both ducts. They appear to serve quite well as estimates of the cutoff frequency due to the finite size of the pressure transducer. Note that the ratio of the transducer diameter to the boundary-layer displacement thickness, d/δ^*, which should be as small as possible for accurate high frequency resolution, was 0.092 in the 8-inch duct as compared to 0.33 for the lowest value found in the literature[14].

The general features and magnitude of the pressure spectral density curves are similar to those observed by Corcos for the turbulent flow in a 1-inch diameter pipe[15]. Over the range of flow velocities used in the panel tests, the ratio of the root-mean-square pressure at the wall to the duct centerline dynamic pressure was approximately 0.009. This is almost twice as large as the value found by Willmarth and Wooldridge for the turbulent boundary layer on a smooth wall[14]. The increase is probably due to the roughness of the acoustic tile-lined walls of the ducts. The ratio of the root-mean-square wall pressure to the average wall shear stress, as calculated from the static pressure gradient along the duct centerline, ranged between 2.8 and 2.0. This range of values is in accord with that found by both Corcos and by Willmarth and Wooldridge.

The longitudinal two-point space-time correlations of the wall pressure measured along the centerline of the 8-inch duct are shown in Figures 5 and 6. For each spacing, Δx, there was an optimum time delay at which the correlation, R, was a maximum. The ratio of Δx to this optimum time delay defines the average speed, U_c, at which the pressure producing eddies were convected downstream. The time delay or τ axis has been multiplied by this average convection speed. If U_c were a constant with Δx, such a coordinate system would produce a pattern in which the locus of the correlation maxima was a straight line, swept at an angle of 45 degrees in the Δx - $U_c\tau$ plane. In practice, this locus was slightly curved in such a way as to indicate that the apparent convection speed increased with increasing transducer separation (Fig. 6).

In general, the variation in U_c with Δx was similar to that reported by Willmarth and Wooldridge[14], the average values of U_c/U being 0.74 in the 8-inch

duct and 0.71 in the 1-inch duct. Correlation of 1/3 octave bandwidth portions of the pressure fluctuations showed that U_c increased with decreasing frequency, suggesting that it is the larger pressure eddies which are convected fastest.

The lateral correlation patterns obtained were unswept, and the correlation pattern for each lateral separation occurred at zero time delay. Measurements of the two-point space correlations ($\tau = 0$) in each duct were hampered by the minimum separation available between the pinhole microphones. However, sufficient information was obtained in the 8-inch duct to indicate that the lateral eddy size was larger than the displacement thickness which, in turn, was larger than the longitudinal eddy size. In addition, the wall pressure field was found to be homogenous in the flow direction and in the vicinity of the duct centerline in the lateral direction, as well as stationary in the statistical sense.

A more complete presentation of the measurements made on the wall pressure field will be published in the near future as part of a UTIA report. It suffices at the moment to point out that the data obtained in the two ducts appear to fall within the general limits of the information available on turbulent boundary layers and, hence, can be taken to represent an approximate model of the wall pressure fluctuations in such a boundary layer.

4.2 Panel Damping

The critical damping ratio, ξ, was measured for numerous, but not all, modes of sample panels mounted in the 8-inch duct. The results presented in Figure 7 are estimated to be accurate to within ±10 per cent at frequencies above 200 cycles/sec and ±25 per cent near the fundamental mode of the panels.

The sharp rise in ξ near the fundamental frequency cannot be attributed to an increase in radiation damping. An estimate of the radiation damping for a 0.008-inch panel at its fundamental frequency was obtained by assuming that the panel acts as a rigid piston in an infinite baffle with an average velocity which has been adjusted to correspond to the mode shape. The calculated value was more than an order of magnitude too small to account for the observed value of the critical damping ratio. It is suggested, as an alternative, that the increase in damping near the fundamental mode may be due to dissipation of energy at the clamped edges of the panel.

At frequencies above 500 cycles/sec the measured damping ratios correspond roughly to those which have been found for steel bars and which are usually considered to be due to hysteresis. Hence, the assumption of constant hysteretic damping and negligible radiation damping may be an adequate approximation when the motion of a panel excited by turbulence is studied, provided that the range of interest is limited to frequencies well above the fundamental mode of vibration and the speed of the convected turbulence is subsonic.

The values of the modal damping used in the theoretical investigation of the panel surface motion were computed from a curve which approximated the experimental critical damping ratios.

4.3 Panel Surface Motion

Figure 8 shows the experimental and theoretical displacement response at the center of a 0.008-inch panel to the turbulent flow in the 8-inch duct. The bulk of the experimental response was in the frequency band between 100 and 1000 cycles/sec. The two large peaks below 50 cycles/sec are shown for the sake of completeness, but it must be considered highly probable that they arose from excitation other than the turbulent flow. The wall pressure measurements indicated that there was a large acoustic standing wave in the duct which completely masked the turbulent pressure fluctuations at frequencies below approximately 50 cycles/sec.

The majority of the observed peaks were successfully associated with the theoretical modal frequencies. There were some deviations which can most likely be attributed to the fact that the narrowest filter available did not allow sharp resolution of the higher order modes and that even the slighest transducer misalignment could produce a response to some unexpected even-order modes or eliminate some expected odd-order modes at the panel center. The displacement frequency content for this panel agreed reasonably well with the upper limit of the response defined by the coincidence condition mentioned previously. This calculated upper limit was approximately 760 cycles/sec. Some additional observations on the upper cutoff frequency will be made in the discussion of the acoustic tests.

When making correlation measurements, the experimental panel response was limited by means of high pass filters to frequencies above 100 cycles/sec, so as to exclude the two suspected peaks in the displacement spectrum. In order to preserve some similarity between the experimental and theoretical models, all peaks below 100 cycles/sec were eliminated from the theoretical response by excluding the low-order modes corresponding to these resonant frequencies from the formulation of the problem. It is this modified theoretical displacement spectrum which is shown in Figure 8 and which is representative of the panel motion used in computing the theoretical space-time correlation of the panel displacement.

In comparing the experimental frequency spectrum of the displacement to the theoretical one, it should be recalled that the theoretical model had simply supported rather than clamped edge conditions. Hence, the resonant frequencies for the real panel and for the theoretical model were different. Moreover, the complete displacement spectrum of the theoretical model had rather large sharp peaks at the allowable resonant frequencies below 100 cycles/sec, except for the fundamental, whereas the experimental results suggest a sharp drop in response below approximately 200 cycles/sec. This sharp drop was accompanied by a broadening of the resonant peaks which was much too pronounced to be attributed to the observed damping. The effect of damping on the width of the resonant peaks can be judged from the theoretical curve which includes the experimentally measured damping for this panel thickness. It is difficult to see how the different edge conditions could account for this difference in the pattern of response.

The experimental correlations of the panel transverse motion are shown in Figures 9 and 10. In these Figures the longitudinal two-point space time correlations, R, are plotted versus spatial separation, Δx, and time delay τ multiplied by the mean convection speed U_c. This presentation is similar to that used for the wall pressure fluctuations. Hence, any running wave content in the panel displacement

will show itself as a pattern whose main features are swept at an angle of approximately 45 degrees in the Δx - $U_c\tau$ plane. The experimental (Figs. 9 and 10) and theoretical (Figs. 9 and 11) correlations are quite similar and both have a pronounced peak and valley running at the proper angle superimposed on a more random pattern.

The consequences of arbitrarily excluding some low-order modes from the theoretical displacement response of the panels were investigated by computing the displacement spectra and correlation patterns for two other effective low cutoff frequencies and for no cutoff. The panel resonant peaks were so sharp (damping so low) that the elimination of some modes below a critical frequency did not seriously change the remainder of the spectrum. For example, the elimination of modes below 100 cycles/sec in the calculation of the spectrum at the panel center changed the displacement at the theoretical modal frequency 113.4 cycles/sec by -0.18 per cent and the displacement at the modal frequency 466.2 cycles/sec by 0.0005 per cent. With regard to the space-time correlations, it was found that the general pattern including the ridge and valley at 45 degrees was retained, but that the period of the damped sinusoidal type oscillations which made up the over-all pattern was heavily dependent on the spectral component with the largest amplitude. The lower the frequency, the more expanded the time scale and vice versa. In addition, it was found that allowing for statistical modal intercoupling as well as increasing the total number of allowable high order modes in the computations produced no changes in the theoretical correlation patterns.

The patterns of the experimental lateral two-point space-time correlation and the filtered longitudinal and lateral correlations were unswept. These results were interpreted as showing that a panel responds mainly in flexural waves running down it in the flow direction and in standing waves in a direction transverse to the flow. The similarity between the space-time correlations obtained with the experimental and theoretical models suggests that an analytical solution using a superposition of modal responses can accurately describe the basic running wave nature of the panel surface motion. However, until the apparent repression of the low order modes can be theoretically accounted for, the method used in this work at least can only be expected to produce qualitative results.

4.4 Panel Noise

A typical 1/3 octave spectrum of the measured sound pressure levels for a 0.0015-inch panel in the 8-inch duct is shown in Figure 12. The background noise and the power level, calculated from the room calibration, are also shown. For all except the thickest panels, the background noise was sufficiently low as to require no correction to the measured sound pressure level over the major portion of the spectra. The total power radiated by each panel was calculated by adding linearly the power levels found in each 1/3 octave bandwidth. In all cases the lower limit of the integration was approximately 100 cycles/sec. Neglecting the power radiated below this frequency leads to an underestimate of the total power, but this error can be shown to be less than 0.5 db for all speeds and panels except those of 0.003-inch thickness. The latter results may be underestimated by as much as 2 db at low speeds.

Several panels of each thickness were tested in the 8-inch duct as it was found that even with very careful mounting there was a certain amount of scatter in the results. Some of this scatter was due to temperature and atmospheric pressure differences. It was not possible to separate the effect of these two quantities, but

it was noted that an increase in atmospheric temperature or density or both increased by as much as 1.5 db the level of the sound power radiated by the same panel on different days. These changes did not appear to affect the shape of the spectrum; however, the shape of the spectrum was changed by slight differences in edge conditions between similar panels. The latter effect was most noticeable in the high frequency response of the panels. The magnitude of the scatter was reduced to an acceptable level by improving the procedure used in mounting the panels in the frame.

From the integrated power spectra one can assess the effect of flow velocity, panel thickness and boundary-layer thickness on the total power radiated. Total power versus flow velocity curves are shown in Figure 13 for all of the 0.002-inch panels tested. The data indicate that for these panels, the radiated power varied approximately as the fifth power of the flow velocity in both the 8- and 1-inch ducts. The average difference in power levels between these panels in the two ducts was 4.5 db. This corresponds to a radiated power which varies as the square root of the boundary-layer thickness, δ , or displacement thickness, δ^* (the two boundary-layer parameters are proportional in this work).

Figure 14 illustrates the effect of panel thickness on the radiated power when flow speed is held constant. The upper line in the figure suggests a variation which is approximately inversely proportional to panel thickness. At lower flow speeds the dependence of radiated power on panel thickness appears to increase. This change in slope of the logarithmic curves with velocity is at least partially due to the necessity of limiting the lowest frequency in the integration of the spectra to 100 cycles/sec. Errors incurred by neglecting frequencies below this value become larger for low flow speeds and thicker panels, tending to make the apparent logarithmic slopes more negative. In partial confirmation of this explanation, note that the average logarithmic slope of the power versus thickness curve for a flow speed of 125 ft/sec becomes approximately the same as that for 170ft/sec if the data points found for the thickest panels are neglected.

Table I summarizes the results obtained for the variation of total power radiated with flow velocity, panel thickness, and boundary layer thickness. Note that the exponent of the flow velocity appears to increase with increasing panel thickness. The low signal-to-noise ratio and small range of flow velocities available with the 0.008-inch panels created some uncertainty as to the proper value of the exponent for this configuration. An argument similar to that used for the thickness variation leads to the conclusion that a variation in sound power proportional to U^6 is most appropriate.

Some measurements of the sound pressure level in the near field of the panels were made for purposes of comparison with the power levels. A progressive increase in magnitude with decreasing frequency of the near field sound pressure level, measured at a distance, z , of 3 inches directly above the center of the panel, when compared to the power level, is shown in Figure 15 for a typical case. Qualitatively, such an effect would be expected from the reduction in radiation efficiency with decreasing frequency for, say, a piston in a large baffle. The relatively close numerical agreement at high frequencies must be regarded as fortuitous, since the near field results depend on the choice of z and the directivity of radiation from the panels. The accentuation of the low frequency end of the spectrum for measurements close to the panel could change the dependence of the near field sound pressure level on the various parameters to something different from that which would be found from measurements of radiated power.

Table I includes the near field results obtained at a distance of 3 inches above the panel center. There is little difference between the two sets of measurements. The largest percentage difference occurred for the dependence on boundary-layer thickness. Presumably, the differences between near field and power level measurements should increase with decreasing distance from the panel. This increasing near field influence has been observed for the variation of over-all near field sound pressure level with flow velocity. A 0.002-inch panel mounted in the larger duct displayed a dependence of the sound pressure level on flow velocity which decreased from $U^{5.3}$ at a microphone distance of 32 inches above the panel to $U^{4.3}$ at a distance of ½ inch above the panel.

4.5 Total Radiated Power-Comparison Between Theory and Experiment

In Table II the variation of the total sound power radiated by the panels with flow velocity, boundary-layer thickness, and panel thickness has been compared to the predictions made as a result of theoretical investigations on the same problem. The analyses of Ribner[2] and of Corcos and Liepmann[6] considered the sound radiated from infinite or at least very large plates. One would expect the results of these analyses to be applicable when the sound radiated by the panels is predominantly at frequencies well above the fundamental mode vibration. This was the case for the panels tested in the present work with the possible exception of the thickest panels, so that a comparison of these theories with the observed data is not unreasonable. Kraichnan[7] considered the radiation from finite square plates, which description fits exactly the experimental panels. The work of Dyer[3] was more directly concerned with the panel surface motion and, as such, has been omitted from this comparison.

Since the theories give different results depending on the combination of the parameters, several answers for the functional dependence of the power radiated on U, δ, and h are possible. The combination of experimental parameters fell between the upper and lower extremes of Ribner's theory, thus both limited cases are presented in the Table. For the remaining two theories, the limiting cases were more easily recognized; thus, only the predictions pertinent to the present tests are included in the Table.

None of the theoretical relationships predicts completely the functional dependence of the radiated power on all three parameters. In general, the theoretical analysis of Corcos and Liepmann gives the most precise analogy with the observed data, although it over-emphasizes the role of boundary-layer thickness. The failure of Kraichnan's very comprehensive work to agree with the current tests appears to lie in his choice of spectrum function for the driving pressures. The assumed spectrum bears little resemblance to the one observed in the ducts. It fits approximately the data measured by Harrison, but it is suspected that the sharp drop in spectral densities above $f\delta^*/U \approx 0.1$ in Harrison's work was caused by transducer size.

Strictly speaking, Ribner's theory holds only for an infinite panel subjected to a 'frozen' convected pattern of turbulence; for subsonic speeds the effects of departure from these assumptions should be least for measurements made in the near field of the actual finite panel. Near-field measurements on the present panels exhibited a velocity dependence approaching U^4 very close to the panel and increased slightly the dependence on boundary layer thickness for measurements moderately close to the panel. This is not in agreement with the U^3 and U^5 variations predicted for the different regimes.

The coincidence condition mentioned previously and used by Ribner and Dyer suggests an upper cutoff to panel radiated sound which is independent of boundary-layer thickness. It is proportional to U^2/h . The experimentally determined upper cutoff frequencies were independent of boundary-layer thickness (see Section 5). They were found to be approximately proportional to U/h for the thinnest panels increasing to U^2/h for the thickest panels.

5. EMPIRICAL SCALING OF SOUND POWER SPECTRA

A typical set of the 1/3 octave power spectra for one panel at various flow speeds is shown in Figure 16. The spectra appear to be limited between a high and a low cutoff frequency, both of which vary with flow speed. In addition, both cutoff frequencies varied with panel thickness at a given flow speed, although not in the same manner. It was found possible to collapse all of the data points for all panels in the 8-inch duct, with the exclusion of points below the lower cutoff frequency, to essentially a single curve. The result is shown in Figure 17. The spectra obtained with panels of 0.002-inch thickness in the 1-inch duct are also included.

The reference thickness, h_0 , and the reference velocity, U_0 , used in this Figure were chosen for convenience. They represented the 1½ thousandths-inch panel thickness and the maximum velocity, 172 ft/sec, attainable in the 8-inch duct. The remaining parameters are P(f) - the power spectral density of the sound radiated in db per cycle, q - the dynamic pressure at the duct centerline, h - panel thickness, U - flow velocity at the duct centerline, and f - frequency.

Before discussing the implications of such an apparently universal representation of the data, it should be emphasized that the low frequency cutoff did not scale with the remainder of the spectrum and that frequencies below this cutoff have been eliminated from the Figure. Uncertainty of the measurements below 200 cycles/sec made it impossible to define a definite relationship between this critical frequency and the parameters which were varied in the tests. It can be said, however, that the low frequency cutoff was well above the fundamental mode of vibration of all of the panels and that it decreased slightly with decreasing flow velocity and increasing panel thickness. Neither the measured panel damping nor a reduction in radiation efficiency with decreasing frequency can be used to account for this phenomenon which was apparent in both the spectra of the acoustic power levels and the panel transverse motion.

Even within this limitation, there are several important points which can be arrived at from the Figure.

First, the upper cutoff frequency was independent of the boundary layer thickness. It is evident from the data that the only effect of boundary layer thickness is to alter the panel response below the upper cutoff frequency. The radiated power spectra followed the shape of the driving pressure spectrum between the upper and lower cutoff frequencies. Note that if the power spectral density of the wall pressure had not leveled off at low frequencies in the 1-inch duct, it is quite possible that the boundary layer thickness would have had no effect at all on the power radiated.

Second, it is necessary to include the panel thickness in the velocity scaling, even for a single panel at various flow speeds. The panel thickness enters the velocity scaling as a modification to the exponent of flow velocity, U. Such an effect might have been expected from the variation in velocity-power laws with thickness, obtained in the over-all sound power measurements (Table I). In addition to this indirect effect, the thickness entered directly into both the power spectral density and the frequency parameters. The combined result was such as to provide an upper cutoff frequency to the power spectra which was approximately proportional to U/h for the thinnest panels, increasing to U^2/h for the thickest panels.

If it is assumed that the entire spectrum, including the low frequencies, collapses when plotted in terms of the generalized parameters, then the functional dependence of the total power radiated on flow velocity and panel thickness can be determined within the limitations of the assumption. This was done simply by integrating the power spectral density over the range of frequencies from zero to infinity in terms of the generalized parameters shown in Figure 17. In order to preserve dimensional consistency in the equation, it was necessary to postulate that the radiated power depends linearly on the panel area. This linear area dependence is untested as panel area was not varied. The final result suggested that the total sound power radiated should be proportional to the inverse of the square root of panel thickness, the square of the dynamic pressure at the duct centerline, and the flow velocity raised to the 4-tenths power of the panel thickness.

The variation of total power radiated with velocity suggested by this investigation was found to be in excellent agreement with the experimentally determined velocity power laws summarized in Table I. Moreover, a U^6 velocity power law is suggested for the previously indeterminate results obtained with the 8 thousandths-inch panels. Such a result tends to confirm the remarks made previously regarding this panel thickness.

The predicted inverse square root variation with panel thickness was not observed; the experimental power varied approximately as the direct inverse of the thickness. It is possible that the difference can be attributed to the assumption made in the analysis that the universal spectrum was valid even below the low frequency cutoff; that is, the effect of the low frequency cutoff was neglected. At higher flow speeds than those attainable in the present tests, it is reasonable to suppose that an inverse square root variation would be approached since the influence of the low frequency cutoff on the total power radiated should be less pronounced.

6. EFFICIENCY OF SOUND GENERATION

The acoustic efficiency of the turbulent boundary layer-flexible panel combination can be defined as the acoustic power radiated divided by the boundary-layer friction power. The boundary-layer friction power was calculated from the average wall shear stress in each duct which in turn was found from the static pressure gradient along the duct centerline. The acoustic efficiencies of the various panels are tabulated in Table III for a Mach number of 0.179. This Mach number represents the upper limit attained in the present tests and the lower limit used in an investigation by Wilson[16] of the sound radiated by a turbulent boundary layer on a rigid wall. The comparable acoustic efficiency found in the rigid wall investigation as well as an estimate made by Lighthill[17] from the available data on jets is shown in the Table for comparison.

It is apparent that for the cited Mach number, panels of the test configuration are much more efficient than either a jet or a turbulent boundary layer flowing over a rigid wall. It is probable that sufficiently thick panels or thin boundary layers at high subsonic speeds would reduce the relative acoustic efficiency of such panels to magnitudes comparable to or even less than that of the other two sound generating mechanisms. However, a rather idealized extrapolation of the data, using the scaling parameters derived from the universal curve of power spectral density, indicated that the acoustic efficiency of panel generated sound is still the largest, although by a lesser amount, for several-fold thicker panels at high subsonic speeds.

7. CONCLUDING REMARKS

A detailed study has been made of the flexural motion and noise generated by thin 11 x 11 inch steel panels flush-mounted in the wall of a turbulent flow channel.

The mean square exciting pressure fluctuation at the wall, its spectral density, and two-point space-time correlations of the pressure were measured with the use of pinhole microphones. By comparison of the results with the information available on turbulent boundary layers, it was concluded that the turbulent channel flow provided at least an approximate model of the wall pressure fluctuations in such a boundary layer.

The flexural response of sample panels was studied by correlation techniques. Relief charts of the experimental two-point space-time correlation versus longitudinal separation and time delay showed pronounced oblique ridges and valleys discernable in a more random pattern. These were interpreted as running waves traveling at a speed equal to the pressure field velocity, superposed on an irregular wave pattern. For comparison, Dyer's idealized theoretical model of panel flexural motion was developed and programmed for a digital computer. The calculated relief plots showed qualitative agreement with the experimental results.

For the thinnest panels over the available range of flow speeds, the experimentally measured sound power varied approximately as the square-root of the boundary layer thickness, the fifth power of the flow velocity and the inverse of the panel thickness. The exponent of the flow velocity increased slightly with panel thickness. The weak dependence of sound power on boundary layer thickness has not been predicted by the theories available at present. Empirical scaling parameters were found which collapse the spectral density data to essentially a single curve for a given boundary-layer thickness.

At a Mach number of 0.179 the panels displayed an acoustic efficiency at least 10 times greater than that of a turbulent boundary layer on a rigid wall and 100 times greater than that of a jet. This margin is reduced but not eliminated on extrapolation to several-fold thicker panels at high subsonic speeds.

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TABLE I

Over-all Sound Power Level and Near Field Sound Pressure Level Relationships

PARAMETER	PANEL THICKNESS h	BOUNDARY LAYER THICKNESS δ	FLOW VELOCITY U				
			8 INCH DUCT				1 INCH DUCT
			h = .0015"	h = .002"	h = .004"	h = .008"	h = .002"
POWER LEVEL	$h^{-1.1}$	$\delta^{0.5}$	$U^{5.0}$	$U^{5.1}$	$U^{5.5}$	U^6 to U^8	$U^{5.1}$
SOUND PRESSURE LEVEL (Z = 3.0")	$h^{-1.0}$	$\delta^{0.8}$	$U^{4.7}$	$U^{4.8}$	$U^{5.3}$	~	$U^{4.8}$

TABLE II

Comparison Between Theory and Experiment for Sound Power Relationships

SOURCE	OVERALL POWER LEVEL RELATIONSHIP	REMARKS
PRESENT EXPERIMENTS	$\overline{p^2} \sim \frac{U^6 \delta^{1/2}}{h}$	Exponent of U increases slightly with h
RIBNER	$\overline{p^2} \sim \frac{U^5 \delta^3}{h^3}$	Low range of U δ/h. For steel panels, this applies when U δ/h < 4 x 10^4 approximately. 1 inch duct results lie in this range.
	$\overline{p^2} \sim \frac{U^3 \delta}{h}$	High range of U δ/h. For steel panels this applies when U δ/h > 1.2 x 10^5 approximately. 8 inch duct measurements lie in this range, except for .008 inch panels.
CORCOS and LIEPMANN	$\overline{p^2} \sim \frac{U^5 \delta}{h}$	For field, thin boundary layer, $L/\delta \ll 1$ $L/\delta \leq .28$ in present tests.
KRAICHNAN	$\overline{p^2} \sim \frac{U^5 \delta^4}{h^3}$	Equation 8.2 of Ref. 7. Thin boundary layer (high frequency sound dominant). Panel high frequency cutoff < high frequency cutoff of driving pressure spectrum. Distributed convection velocity, low constant damping $\left[\frac{h}{2L}\frac{2u_p}{U_c}\right]^2 \ll \frac{1}{8}$; u_p 5000 fps for steel $\left[\frac{h}{2L}\frac{2u_p}{U_c}\right]^2 \leq 5 \times 10^{-3}$ in present tests

TABLE III

Acoustic Efficiency at M = .179

TYPE OF MECHANISM	ACOUSTIC EFFICIENCY	
FLEXIBLE PANEL EXCITED BY TURBULENT BOUNDARY LAYER (PRESENT WORK)	$.93 \times 10^{-5}$	.0015" PANEL, 8" DUCT
	$.84 \times 10^{-5}$	.002" PANEL, 8" DUCT
	$.37 \times 10^{-5}$	.004" PANEL, 8" DUCT
	$.23 \times 10^{-5}$	.002" PANEL, 1" DUCT
TURBULENT BOUNDARY LAYER ON A RIGID WALL (REF.16)	$.14 \times 10^{-6}$	
JET (REF.17)	$.11 \times 10^{-7}$	

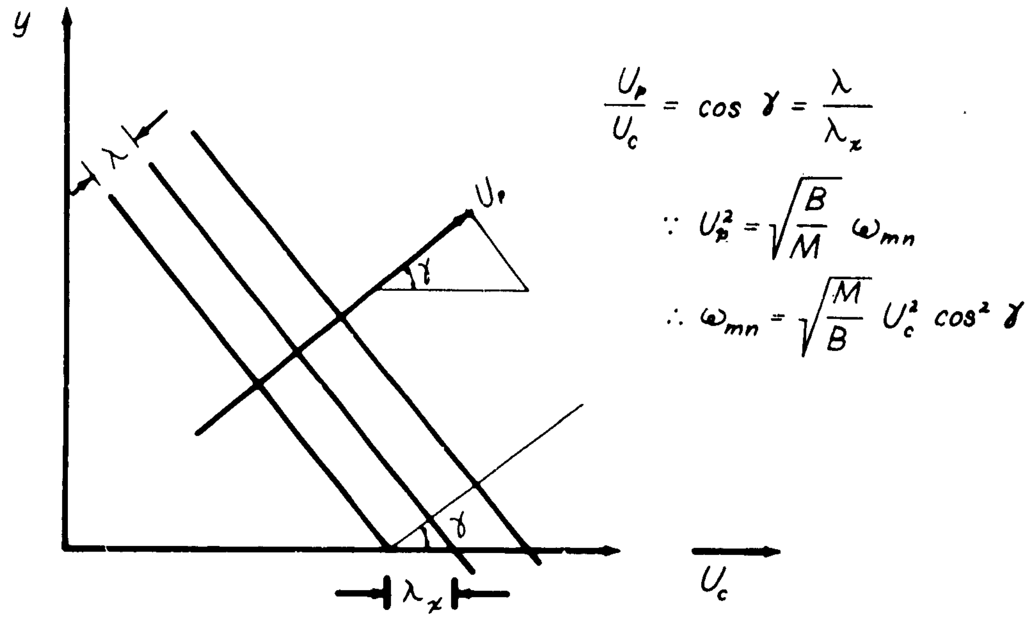

Fig. 1 Pressure and flexural wave matching process at coincidence

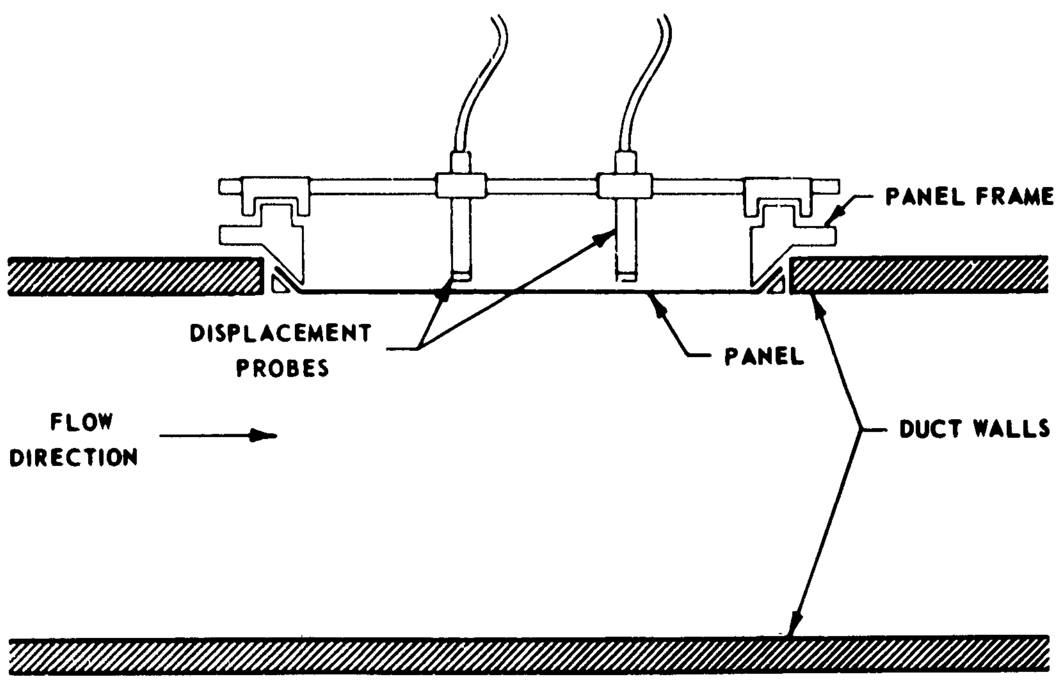

Fig. 2 UTIA low noise air duct with panel in position

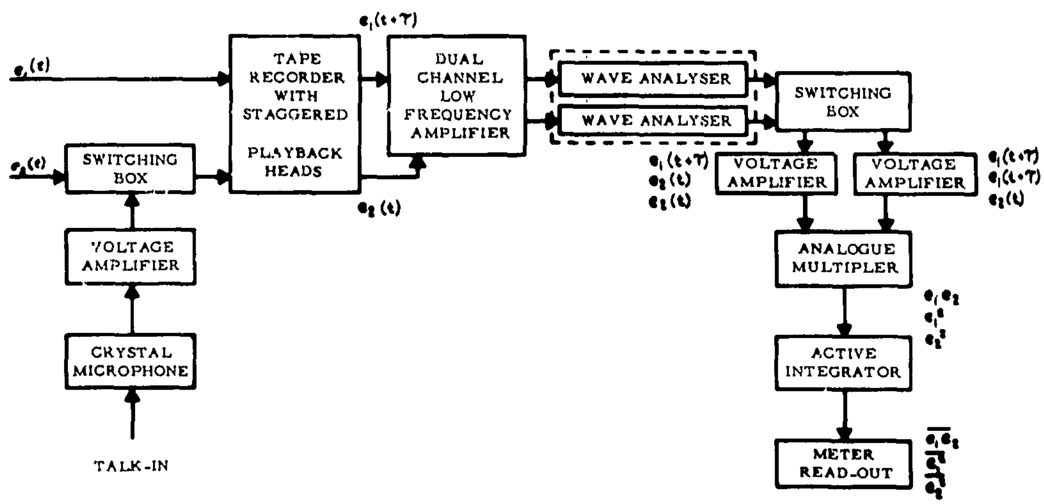

Fig. 3 Electronic correlator

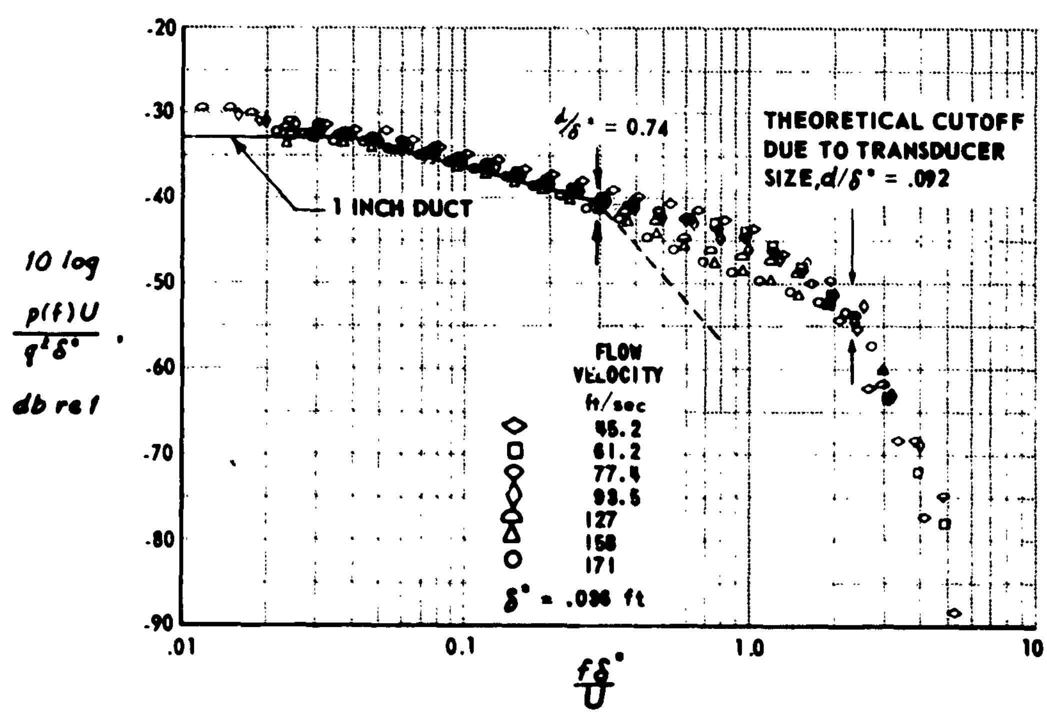

Fig. 4 Spectral density of wall pressure fluctuations (8 inch duct)

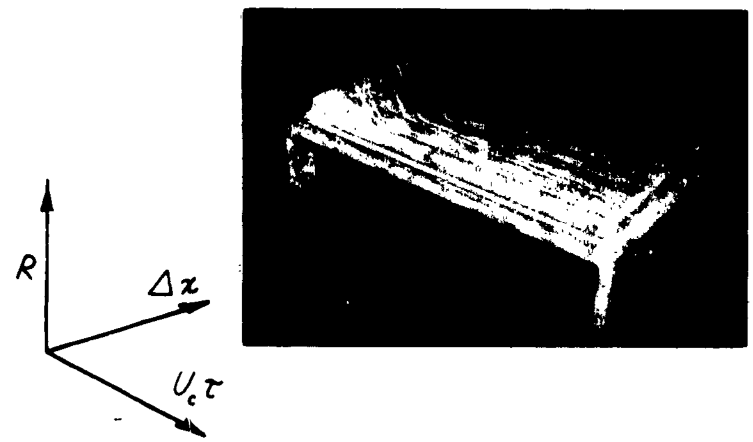

Fig. 5 Experimental two-point space-time correlations of wall pressure (flow speed 170 ft/sec, channel depth 8 in.)

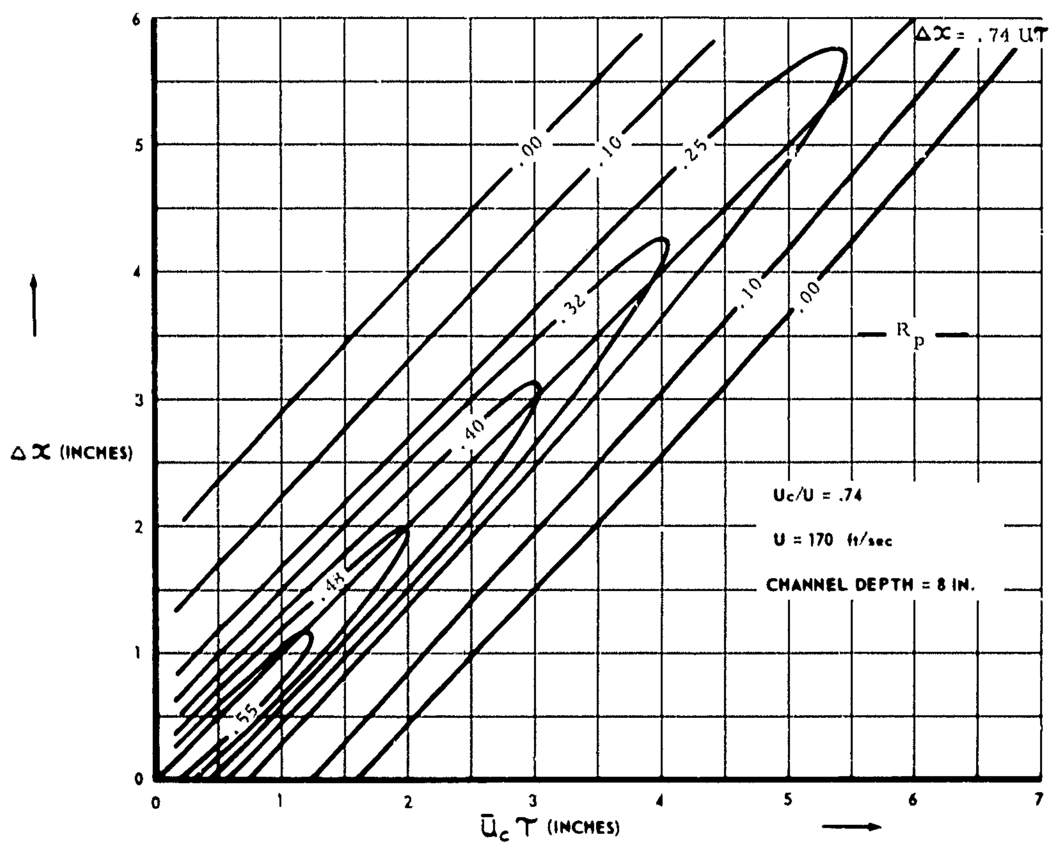

Fig. 6 Lines of constant longitudinal correlation: wall pressure fluctuations

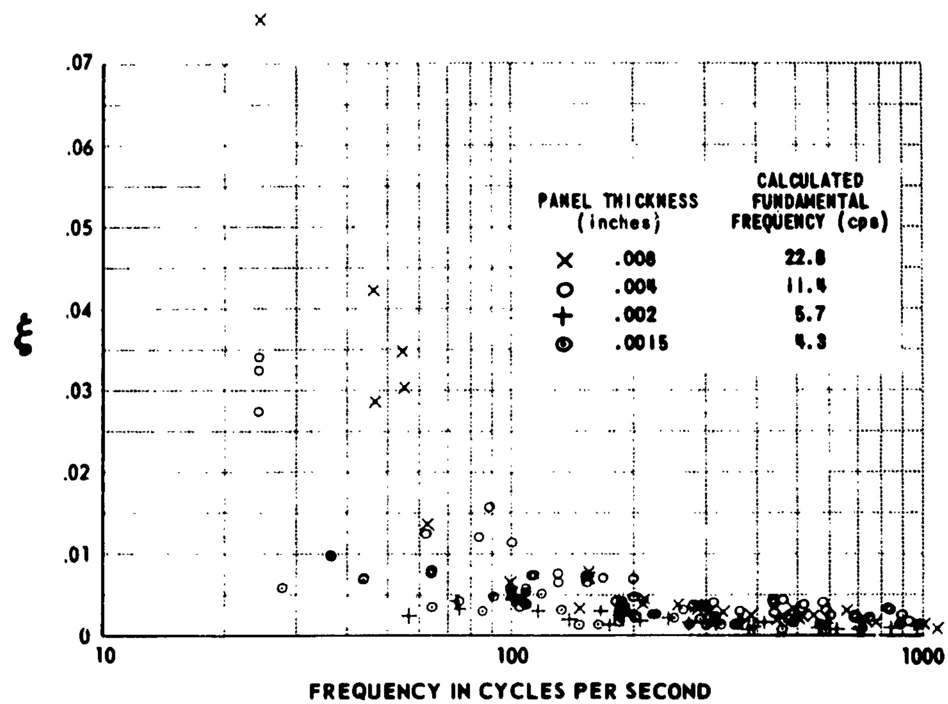

Fig. 7 Experimental damping ratio of panels mounted in 8 inch duct

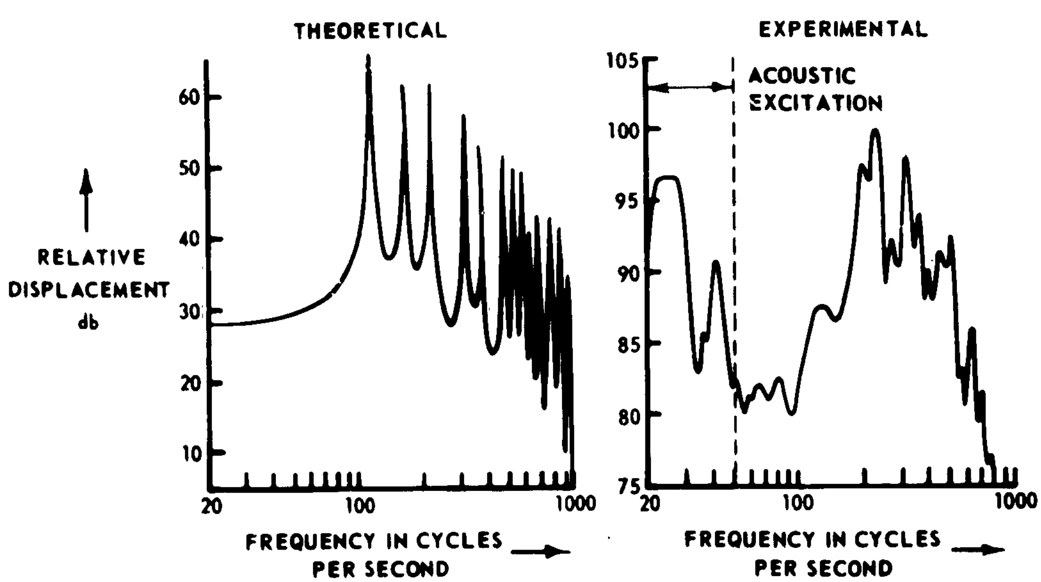

Fig. 8 Spectrum of transverse surface motion at panel center (flow speed 170 ft/sec, channel depth 8 in., panel thickness 0.008 in.)

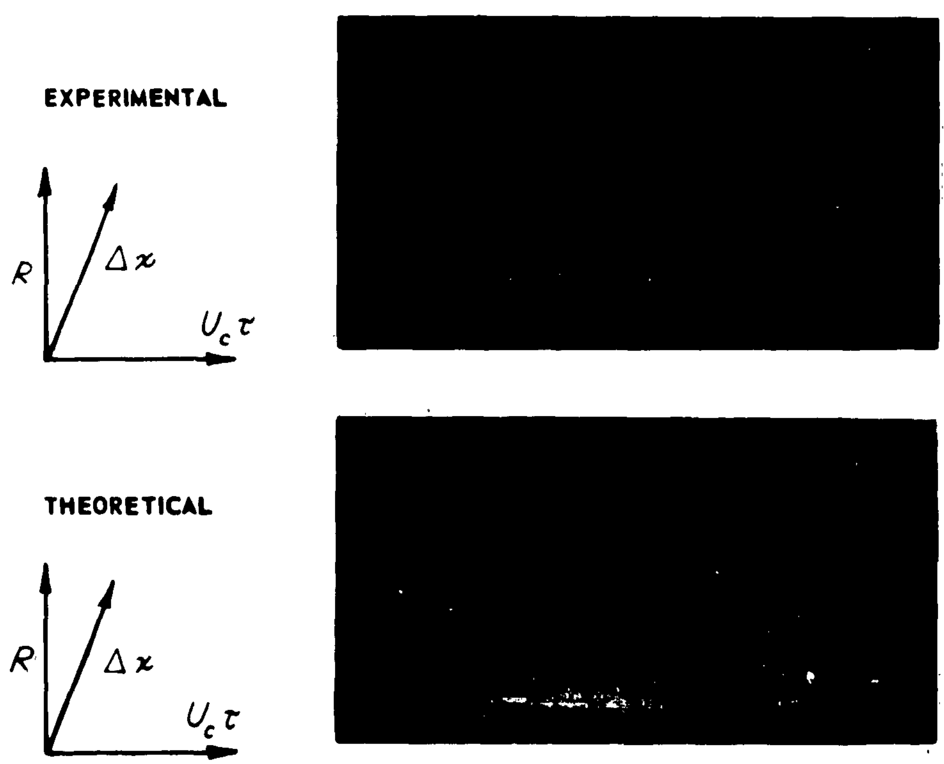

Fig. 9 Two-point space-time correlations of panel transverse motion (flow speed 170 ft/sec, channel depth 8 in., panel thickness 0.008 in.)

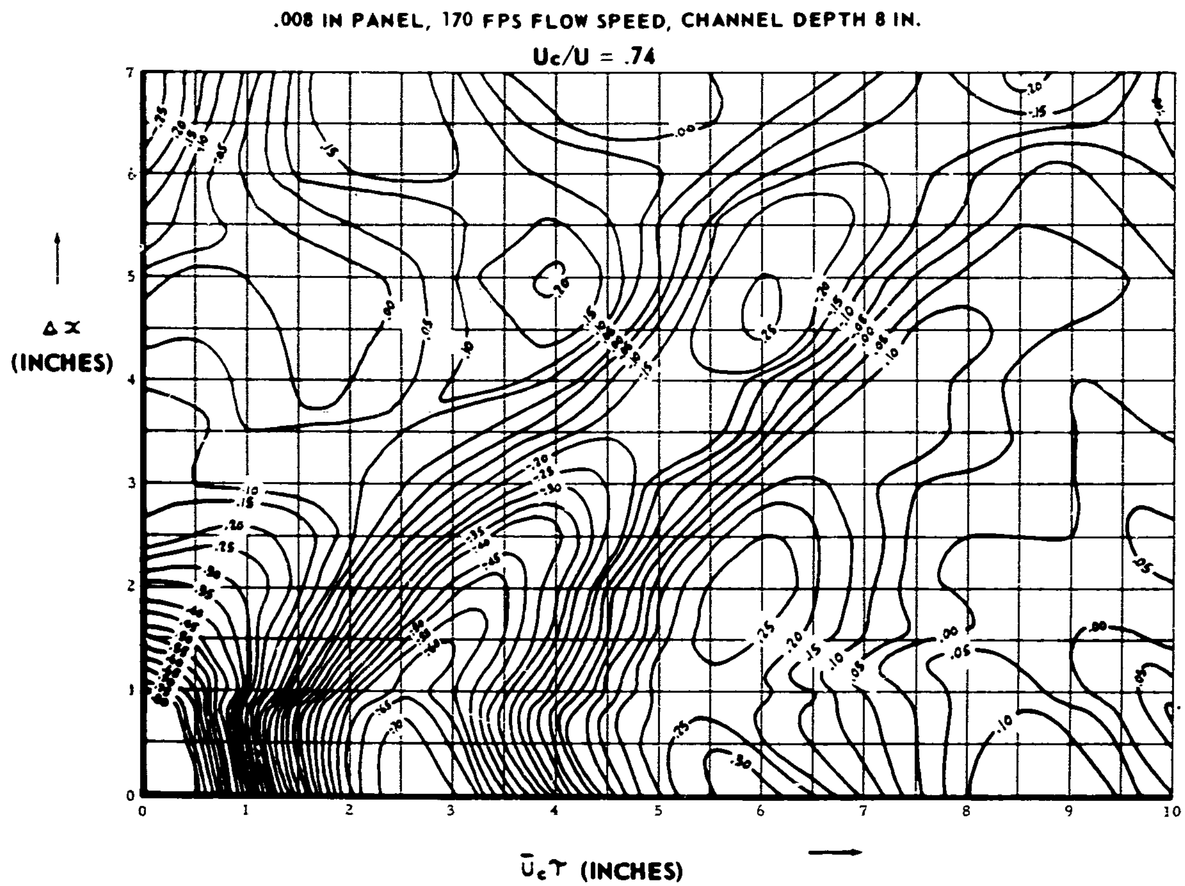

Fig. 10 Lines of constant longitudinal correlation panel surface motion (experimental)

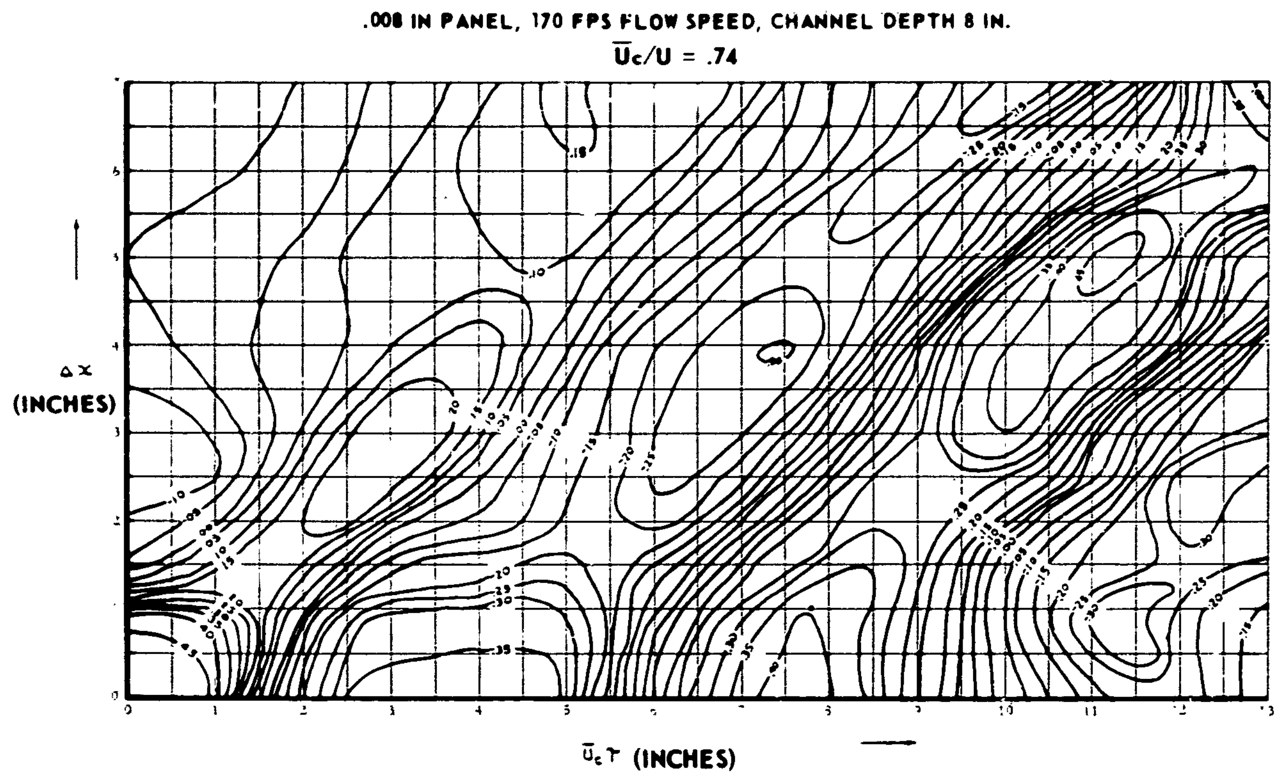

Fig. 11 Lines of constant longitudinal correlation panel surface motion (theoretical)

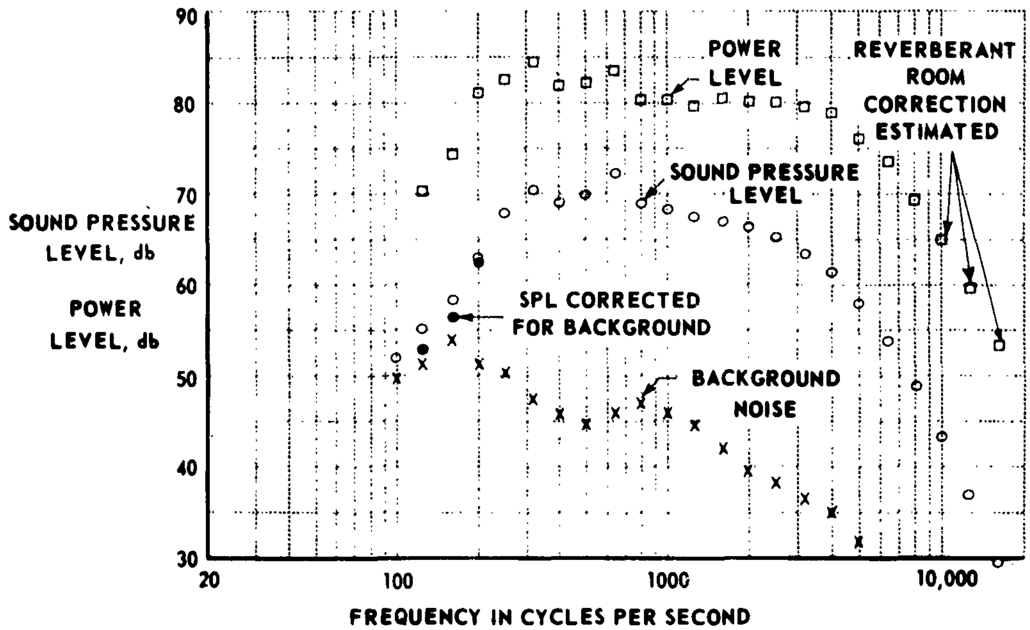

Fig. 12 1/3 octave spectrum of sound radiated by a 0.0015 inch panel (channel depth 8 in., flow speed 171 ft/sec)

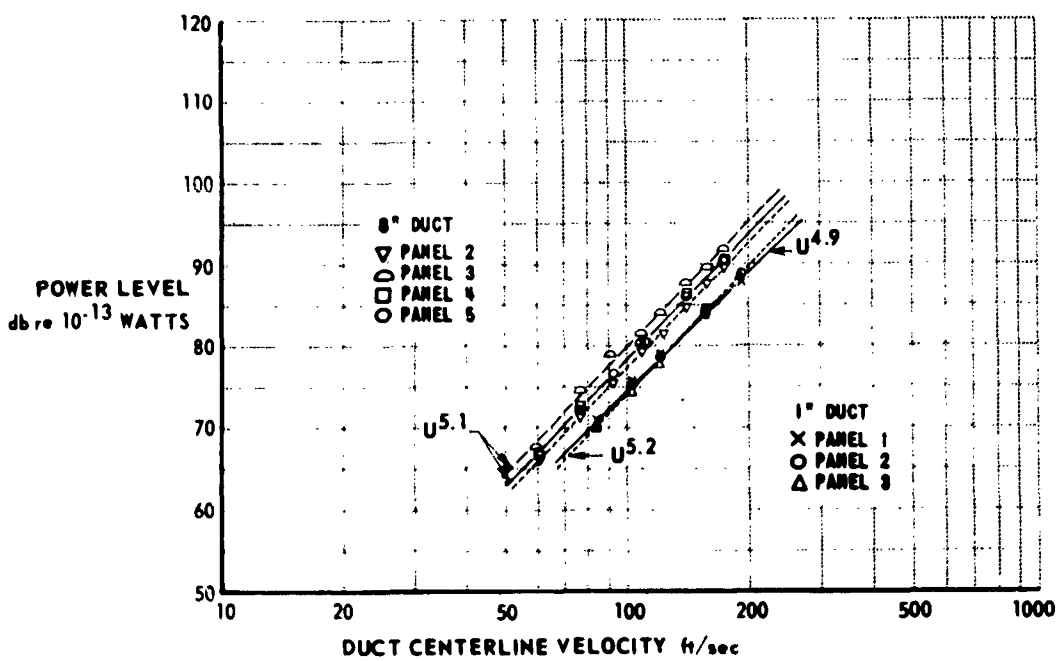

Fig. 13 Total sound power radiated versus velocity (panel thickness 0.002 in.)

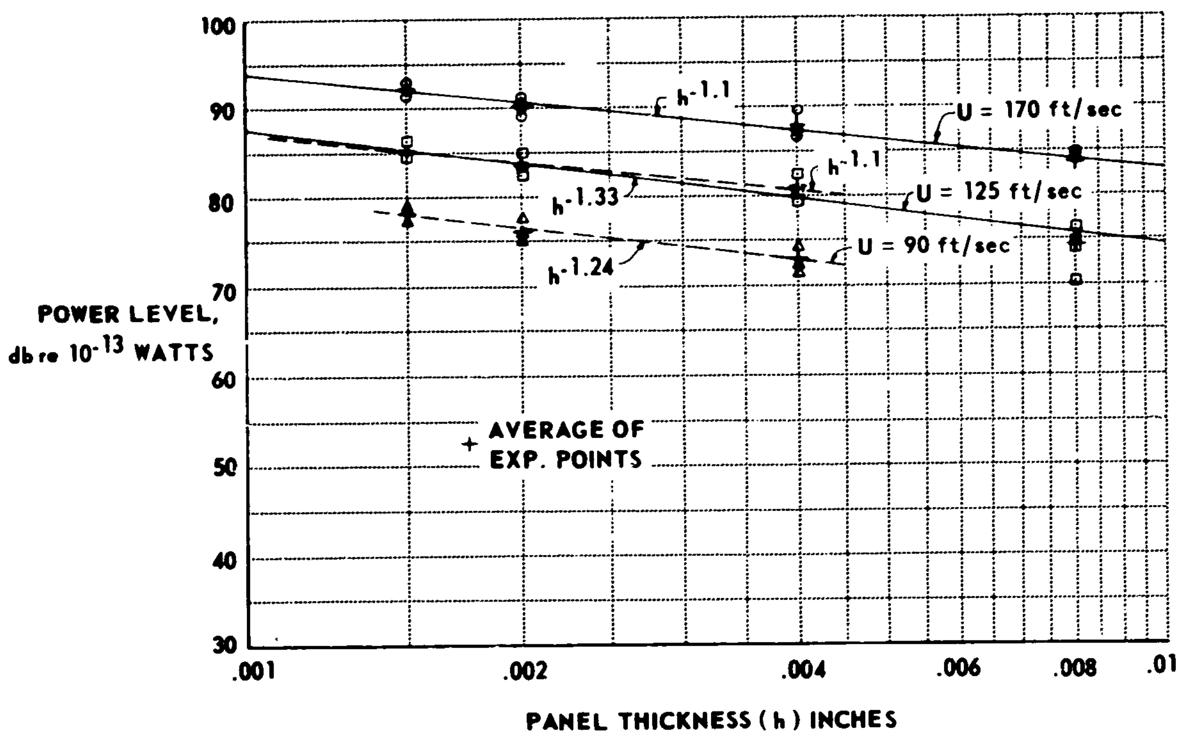

Fig. 14 Total sound power radiated versus panel thickness (channel depth 8 in.)

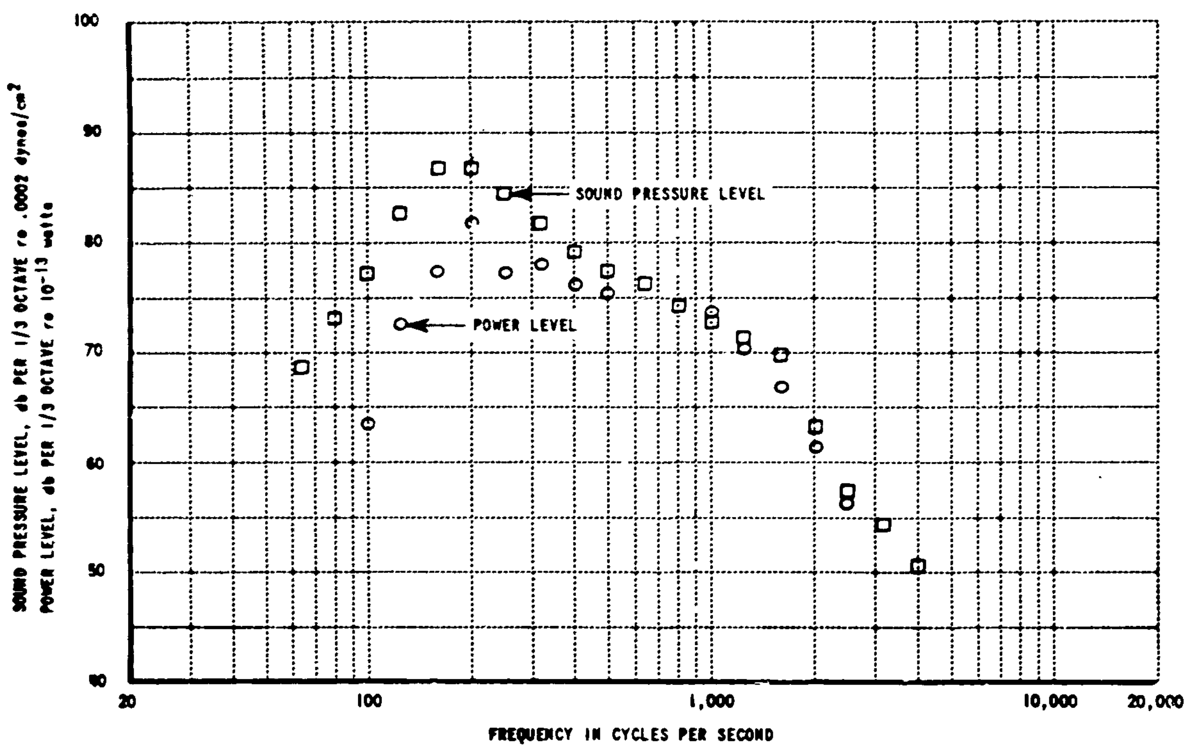

Fig. 15 Comparison of near field (z = 3.0 in.) sound pressure level spectrum with power level spectrum for a 0.004 inch panel in 8 inch duct (bandwidth = 1/3 octave, flow speed = 172 ft/sec)

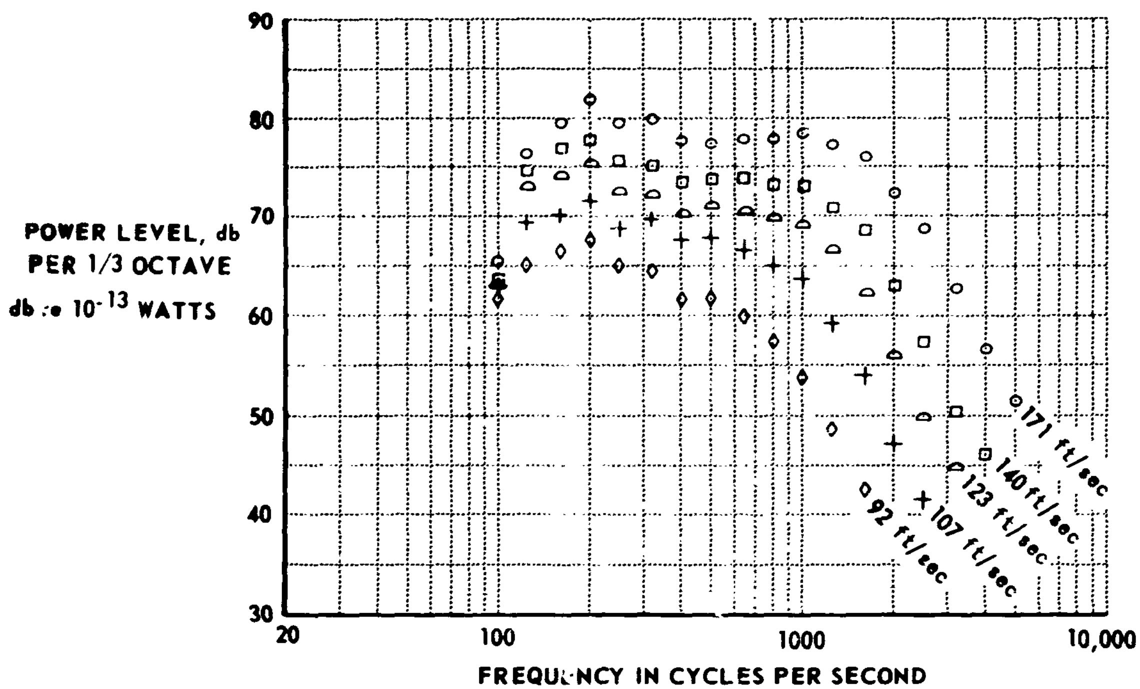

Fig. 16 1/3 octave bandwidth spectra of sound radiated by a 0.004 inch panel in 8 inch duct

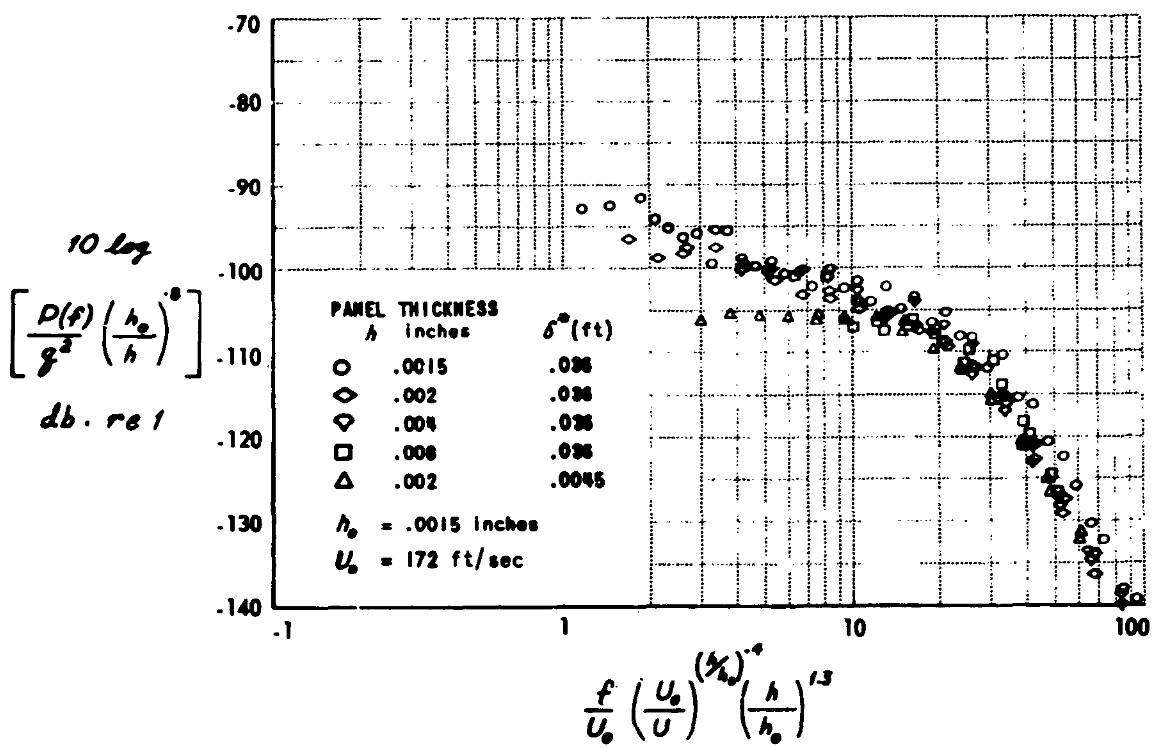

Fig. 17 Universal curve of sound power spectral density. (Data shown include results for the highest and lowest flow speeds at which each panel was tested. Frequencies below low frequency cutoff have been excluded)

DISCUSSION

Comment by J.A. Maurin

Ce ne sont pas des modes fondamentaux de vibration du panneau qui sont en jeu, mais des harmoniques très élevés.

Authors' reply

Oui.

G.M. Corcos

It seems that the acoustic spectrum is almost free of resonance peaks, does it not? I seem to remember that Paul Weyer at the California Institute of Technology observed the same thing on the outside of a vibrating sleeve around a turbulent pipe flow.

Authors' reply

The acoustic spectra were measured only with filters of 1/3 octave bandwidth. The effective power spectrum level of the sound radiated was calculated from the 1/3 octave analysis by correcting for the filter bandwidth. Such a method obscures any resonant peaks in the response if they are close together. The minimum bandwidth of 1/3 octave was necessary in order to use the reverberant room technique for obtaining power level. The displacement spectra on the other hand were measured with an effective bandwidth of $5\frac{1}{4}$ per cent for most of the frequency range and $1\frac{1}{8}$ per cent when finer detail was required. The latter spectra showed a definitely peaky response even at high frequencies.

M. Strasberg

Did you try driving the panel mechanically at various frequencies to determine whether the panel response, when excited mechanically at a point, showed the broadening of the resonant responses as observed when excited by the turbulence? I am thinking that a possible cause of the broadening may be due to coupling between mechanical modes of vibration, a coupling which was assumed not to exist in the theoretical analysis.

Authors' reply

The damping measurements, which were made by measuring the sharpness of the resonant peaks in the displacement response when the panel was excited at a point by an electromagnet, can be referred to in answer to your question. At the frequencies where the broadening of the peaks in the panel response to turbulence were apparent (between 60 and 200 cycles/sec), the damping measurements indicate that the peaks should be only 1 to 3 cycles/sec wide at the half power points if damping were of major importance in determining the width of the peaks. I do not recall observing any evidence of modal intercoupling when making the damping measurements in this range of frquencies.

Actually some theoretical calculations were made which allowed for statistical modal intercoupling. The results showed no significant differences from the calculations in which modal intercoupling was assumed to be nonexistent.

A. Powell

Could there be an error with regard to the scales of Figure 8? How do the experimental and theoretical values compare?

Reply by G.R. Ludwig

Perhaps I did not explain this Figure very clearly. The numbers shown on the coordinates for relative displacement cannot be compared directly. The reference level for the experimental curve was arbitrary and gives no indication of the absolute displacement amplitude. The numerical values of the overall root-mean-square displacement for the particular configuration shown in Figure 8 were .00014 inch experimentally. and 0.00062 inch theoretically. (If the theoretical modal response below 100 cycles/sec is not excluded, the theoretical r.m.s. displacement becomes .00195 inch.)

G.C.C. Smith

(i) How did you define, develop and achieve satisfactory panel mounting conditions? Have you compared mode spectra and/or shapes with theoretical modes?

(ii) Did your displacement measurements include mode with streamwise panel center line modes?

(iii) Did you consider the use of a non-reverberant receiving chamber?

(iv) Did you investigate the behavior of mode damping with amplitude, and were the dampings quoted for displacements comparable with those obtained in the noise tests? I ask this because the radiated power will obviously be dependent on structural damping, hence any non-linear damping characteristics are of interest.

Authors' reply

(i) The definition of satisfactory mounting was based mainly on the requirement of obtaining repeatable results. Preliminary tests indicated that a panel mounted with wrinkles at the edges or which contained an unknown amount of tension could not be duplicated. The results were very sensitive to these two conditions. The development of the mounting procedure was primarily one of developing a reliable method of bending the edges of the panel before clamping it in the frame and then taking care to avoid wrinkles when tightening the clamps. The only check on the final result was visual, but even this simple check usually rejected one out of every two attempts at mounting the panels.

(ii) Displacement spectra were measured only at the panel center. Overall r.m.s. displacement was measured along the panel centerline (in the flow direction)

at distances of .375, 2.375, and 5.5 inches from the leading edge, and also at a distance of 2 inches from the leading edge and 3 inches from one side. Correlation measurements of displacement were made along the panel centerline in the streamwise direction and in the lateral direction.

(iii) No, we did not. The choice was made for two reasons: (1) the reverberant room was available, and (2) the number of measurements required to obtain radiated power by any other method would be prohibitive.

(iv) Damping measurements were checked at several different levels of excitation and no evidence of non-linearity was observed. Hence the damping data presented is for linear vibration. Unfortunately the absolute calibration of the displacement probe was not available at the time the damping measurements were made, so I do not know whether the amplitudes were comparable to the amplitude of the response to turbulence. However, I feel that the panel response to turbulence was also in the linear range as the measured overall r.m.s. displacement at the panel center was less then the panel thickness.

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M.Y. el Baroudi, G.R. Ludwig and H.S. Ribner
1963
33 pp., incl. 17 refs., 17 figs & discussion

A detailed study has been made of the flexural motion and noise generated by 11 × 11 inch steel panels flush-mounted in the wall of a turbulent flow channel. The mean square exciting pressure fluctuation at the wall, its spectral density, and two-point correlations of the pressure were measured with the use of pinhole microhpnes.

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AN EXPERIMENTAL INVESTIGATION OF TURBULENCE-EXCITED PANEL VIBRATION AND NOISE (BOUNDARY-LAYER NOISE)
M.Y el Baroudi, G.R. Ludwig and H.S. Ribner
1963
33 pp., incl. 17 refs., 17 figs & discussion

A detailed study has been made of the flexural motion and noise generated by 11 × 11 inch steel panels flush-mounted in the wall of a turbulent flow channel. The mean square exciting pressure fluctuation at the wall, its spectral density, and two-point correlations of the pressure were measured with the use of pinhole microphones.

The flexural response of sample panels was studied by correlation techniques. The calculated relief plot of correlation shows qualitative agreement with the experimental results.

534.121.4:532.526.4

1c5:3b2f

This Report is one in the Series 448-469, inclusive, presenting papers, with discussions, given at the AGARD Specialists' Meeting on 'The Mechanism of Noise Generation in Turbulent Flow' at the Training Center for Experimental Aerodynamics, Rhode-Saint-Genèse, Belgium, 1-5 April 1963, sponsored by the AGARD Fluid Dynamics Panel.

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